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Performance of Piston Pin-Bearings Under Low Suction Pressure Conditions

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ABSTRACT

Hydrodynamic analysis is used to study the effect of lowering the suction pressure while holding the discharge pressure constant on piston pin bearing performance. It is shown that although this causes bearing loads to increase slightly, the major effect is on the piston-pin unloading behavior.

INTRODUCTION

A restriction in the expansion device of a piece of refrigeration equipment can cause the suction pressure to decrease while the discharge pressure remains relatively constant. This is because the mass of refrigerant on the low side will fall until saturation conditions (i.e. liquid and gas are both present) in the evaporator are no longer possible and the suction pressure can then fall below the saturation pressure for the evaporator temperature. Conversely the mass in the high side will increase, but not enough to make it all liquid, so saturation conditions will remain and the discharge pressure will tend to remain at the saturation pressure for the condenser temperature. This analytical study attempts to show the effect of this on piston pin bearing performance.

GOVERNING EQUATION

The partial differential equation that governs hydrodynamic lubrication is the Reynolds equation, which for an incompressible lubricant, dynamically loaded finite bearing is given (Lloyd, T., Horsnell, R. and McCallion, H., 1966) as follows:

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\eta}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{\eta}\frac{\partial p}{\partial z}\right)=6u\frac{\partial h}{\partial x}+12\frac{\partial h}{\partial t} \qquad (1)$$

where p is the pressure, h is the film thickness, u is the surface velocity, x is the distance around the bearing, z is the distance across the bearing and n is the viscosity of the oil. Given h, the film thickness as a function of x, z and t, the pressure distribution p as function of x and z at any time instant can be found from the solution to this equation. The pressure distribution can then be integrated to give load, friction moment and oil flow.

Numerical Procedure

The Reynolds equation for a finite bearing has no exact solution. So a finite difference method is used to solve for the pressure distribution numerically. Gauss-Seidel-Newton relaxation method was used to calculate the pressure at each discrete node.

RESULTS AND DISCUSSION

The analysis was done for a discharge pressure of 1180.36 kPa (171.2 psig) and 6 different suction pressures shown in Table 1. The various parameters that were generated along with the pressure distribution are also shown in the table. Figures 1 to 4 show the maximum load, minimum film thickness, average oil flow rate and degree of unloading plotted for the various suction pressures. The following can be observed from these plots.

1. The maximum load on the bearing increases with decrease in suction pressure but levels off for very low suction pressures.
2. The minimum film thickness drops initially (over 21% as the suction pressure changes from positive to negative values) but levels off for very low suction pressures.
3. The average oil flow rate and the degree of unloading decreases and becomes zero asymptotically for very low suction pressures.

It was also observed that the number of cycles for the convergence of the routine increased with a decrease in suction pressure. A comparison of the plots of the Journal Center Locus (Figure 5) shows that the journal center moves outward with decrease in suction pressure. Also from Figure 6, it can be seen that the region of unloading slowly decreases and becomes zero below a suction pressure of 51.20 kPa (–7.27 psig).

CONCLUSIONS

Though restriction in a capillary tube or expansion valve will cause piston pin bearing loads to go up slightly, the major effect is on the piston pin unloading behavior. As suction pressure decreases, the force reversal during the re-expansion portion of the cycle diminishes. This reduces the tendency for the journal to move back and away from the loaded side of the bearing. The result is smaller oil film thickness and lower oils flow rates.

This analysis does not attempt to model the loss of cooling flow that will also result from a restricted expansion device. It is well known that loss of cooling flow will cause internal temperatures in the compressor to climb, which in turn will have a detrimental effect on oil viscosity. If internal temperature climb too high, breakdown of the oil, as well as motor wire insulation will occur.

ACKNOWLEDGEMENT

I would like to thank James R. Lenz, Mechanical Engineer, Tecumseh Research Laboratory, for his help in publishing this paper.

REFERENCES

Lloyd, T., Horsnell, R. and McCallion, H., 1966, "An investigation into the preformance of dynamically loaded journal bearings: Theory", Proceedings of the Institution of Mechanical Engineers, Vol. 181, pp. 1- 8.

Shigley, J. E., "Mechanical Engineering Design", McGraw-Hill book company, 1972.

Table 1. Parameter values for the various suction pressures

Suction Pressure (kPa)	377.45	228.32	114.63	51.20	19.97	8.25
Maximum Load (N)	191.21	219.02	239.66	250.67	252.85	255.33
Min. Film Thickness (mm)	7.03E-04	6.13E-04	4.80E-04	1.56E-04	1.69E-04	2.43E-04
Avg. Oil Flow Rate (mm^3/sec)	1.8416	1.4090	0.7405	0.2079	0.0912	0.0877
Degree of Unloading (%)	11.84	8.97	4.41	0.47	0.38	0.44
Average Load (N)	57.96	57.47	51.41	42.25	31.10	24.02
Power Loss (Watt)	0.0070	0.0075	0.0087	0.0157	0.0154	0.0130

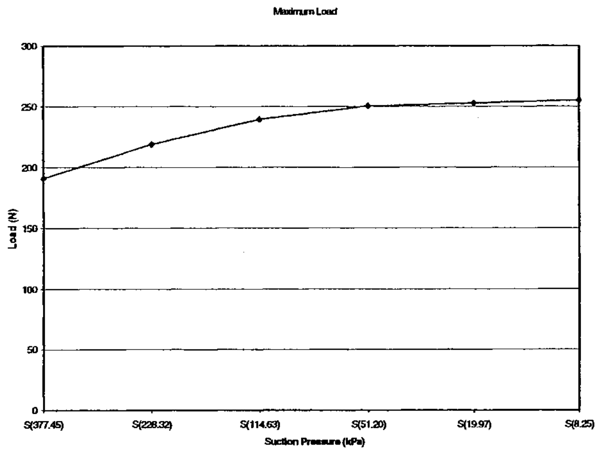


Figure 1. Maximum Load

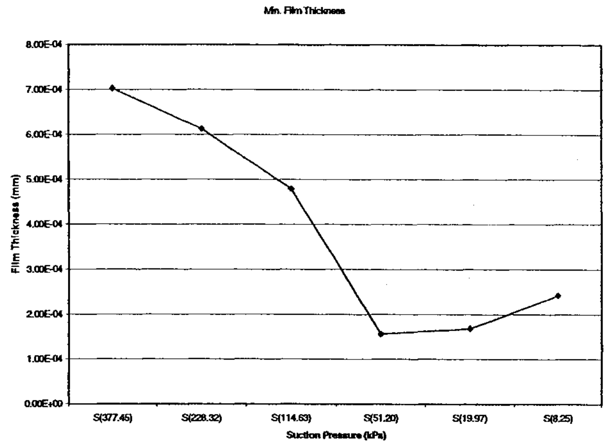


Figure 2. Minimum Film Thickness

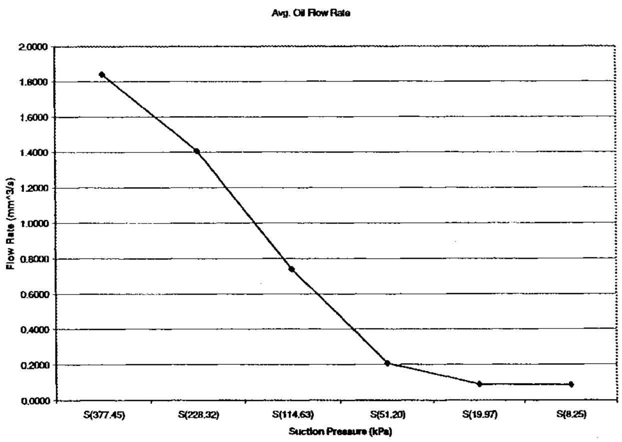


Figure 3. Average Oil Flow Rate

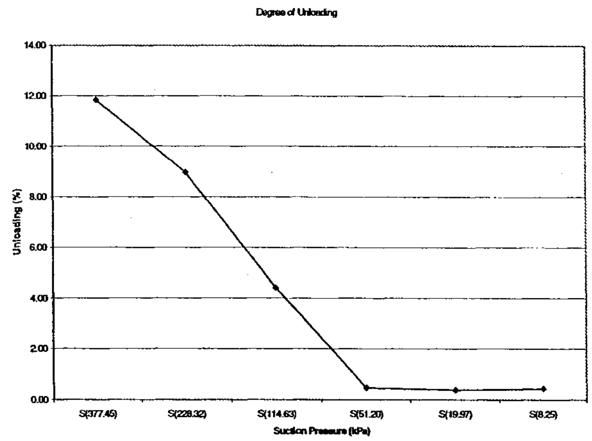
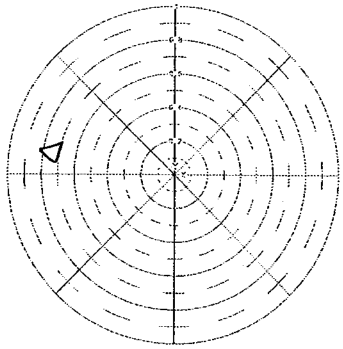
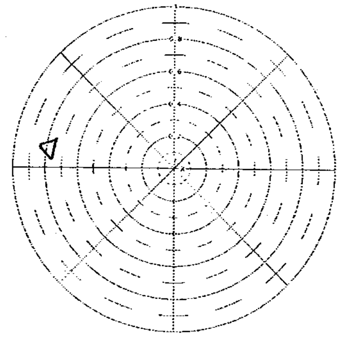


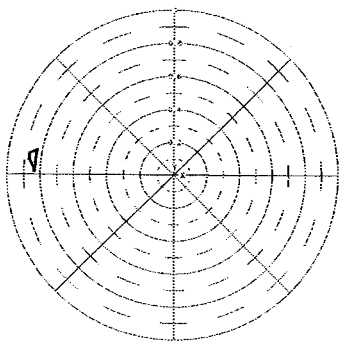
Figure 4. Degree of Unloading



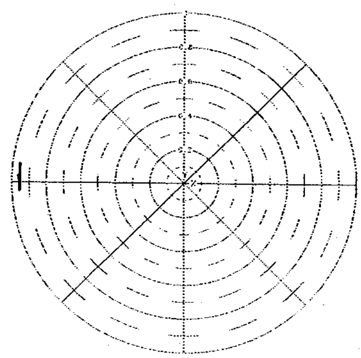
(a) SP - 377.45 kPa (40.05 psig)



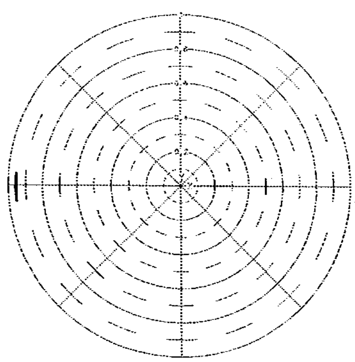
(b) SP - 228.32 kPa (18.42 psig)



(c) SP - 114.63 kPa (1.93 psig)

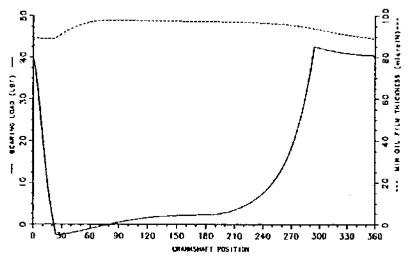


(d) SP - 51.20 kPa (-7.27 psig)



(e) SP - 19.97 kPa (-11.80 psig)

Figure 5. Journal Center Locus For Various Suction Pressures.



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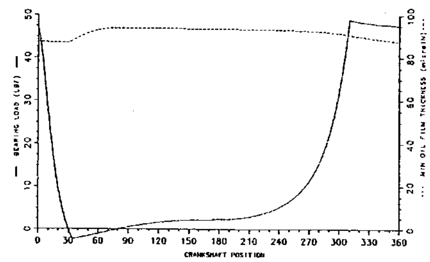

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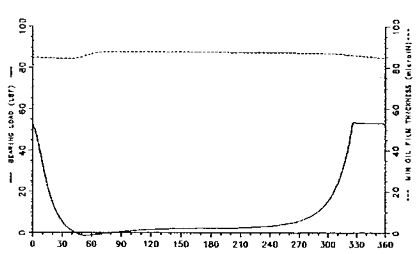
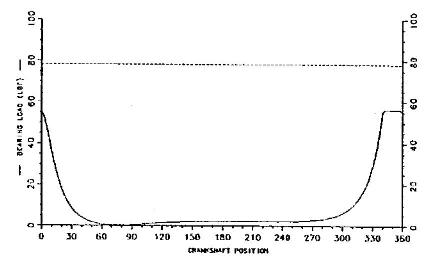


(c) SP – 114.63 kPa (1.93 psig)



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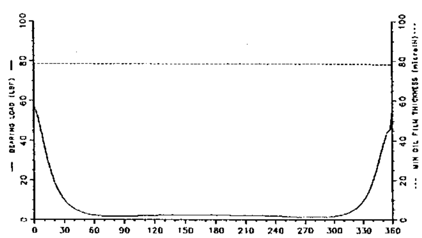

(e) SP – 19.97 kPa (-11.80 psig)

Figure 6. Bearing Load And Minimum Film Thickness For Various Suction Pressures.